

Advanced condition monitoring of Pelton turbines

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Abstract – The ability of hydropower to adapt the electricity generation to the demand is necessary to integrate wind and solar energy into the electrical grid. Nowadays hydropower turbines are required to work under harsher operating conditions and an advanced condition monitoring to detect damage is crucial.

In this paper the methodology to improve the condition monitoring of Pelton turbines is presented. First, the field data obtained from the vibration monitoring of 28 different Pelton turbines over 25 years has been studied. The main types of damage found were due to fatigue, cavitation and silt erosion. By analyzing the vibration signatures before and after maintenance tasks, the symptoms of damage detected from the measuring locations were determined for each case.

Second, a physics-based model using numerical methods (FEM) was built-up in order to simulate the dynamic behavior of the turbine. The model was validated with the results obtained from on-site tests that were carried out in an existing turbine. The deformations and the stresses of the runner under different operating conditions could be then computed.

The calibrated model was used to analyze in detail the effect of misalignment between nozzle and runner. In historic cases, this abnormal operating condition lead to severe damage in the turbine, due to the effect of fatigue in some locations of the buckets. The model reproduced rather well the symptoms detected in the field measurements. The stresses could be calculated which eventually can be used to estimate the remaining useful life of the turbine.

I. INTRODUCTION

Hydropower is regarded as one of the most important sources of energy in sustainable power generation. The biggest share of worldwide electricity production from renewable energies comes from hydroelectricity [1], which also retains one of the highest efficiencies. Hydro utilities are also known for their regulation capacity: power plant operators are capable of adjusting the power outcome depending on the electricity demand. With the implementation of new renewable energies, such as wind and solar, there is an increasing percentage of power that is tied to the variability of its sources, and thus cannot be adjusted to the demand. As electricity production has to match demand at any time, hydropower has become a key player to keep the grid stability [2]. In order to adapt to this new scenario, hydro utility companies have been progressively switching from baseload to a more flexible power production.

The Pelton turbine stands as one of the main types of turbines used in hydropower. This was patented in 1880 by the American inventor Lester Allan Pelton and it is the only impulse turbine used for high power generation. Pelton turbines are generally used in locations where head is higher than 500m (the head is the difference in elevation between the upper reservoir level and the turbine level) and represent around 20% of total installed hydropower turbines. The most powerful Pelton turbines in the world are located in Bieudron, Switzerland. The Bieudron Power Station features three Pelton turbine units with a diameter of 3.993m, rated power per unit of 423MW and a rated head of 1.883m [3]. While low/middle head turbines (Francis and Kaplan) have been extensively exploited, high head hydropower still waits for its full development, especially in some countries like China. At present, the research on dynamic problems and silt erosion is of great interest to increase the design head and to reduce maintenance.

A Pelton turbine is an action-type turbine. This means that the mechanical power is only obtained from a change in the kinetic energy of the water. In the power plant, one or more nozzles (called injectors) expel a high-speed water jet that impulses the runner. Pelton turbine plants offer some advantages over those of reaction-type turbines. On the one hand, they have a large regulation capacity: power can be regulated from 10% to 100% of total capacity with a maximum efficiency of 92%. This is adjusted by means of a sharp-pointed piece (needle) placed inside the injector that can be easily controlled by a servomotor to regulate discharge. In terms of maintenance, the disassembling and inspection of a Pelton turbine is carried out with relative ease compared with a Francis or a Kaplan turbine, due to its simpler installation [4]. In Figure 1 a horizontal shaft Pelton turbine ready for inspection is shown.

Figure 1. Pelton turbine open for inspection. The turbine runner and the electrical generator can be seen.

The runner of a Pelton turbine basically consists of a disk whose rim is surrounded by several buckets. These buckets are designed to receive the impact of the high-speed water jet coming out of the injectors. The inner surface of the buckets is molded as to split the jet into two and divert it almost 180°, thus taking advantage of all the water's kinetic energy. As a result, a

strong force is applied to the bucket (and in the tangential direction of the runner), and maximum torque is transmitted to the shaft. Figure 2 shows a picture and a sketch of the buckets of a Pelton runner with the injector and the needle. The attachment of the bucket to the disk is, due to its cantilever structure, the area that bears the highest stress values in a Pelton runner. In addition, each bucket receives the impact of the water jet several times during operation, which depending on the number of nozzles can range from one to six times per revolution. In the long term, the periodic stresses at the base of the bucket (and other locations of the bucket) end up damaging the structure due to the fatigue of the material. After some time of operation, the material may present some cracks. If these are untreated, the turbine is prone to suffer serious damage in the future. In order to avoid expensive disassembling and reparation costs, a periodic monitoring of the machine is mandatory.

Figure 2. Pelton turbine showing injector and buckets

There are different procedures to evaluate the state of a turbine. Historically, periodic inspections were carried out to check the structural condition of the machine [5]. However, the costs of downtime and disassembling were high, and the possibility of preventing damage was very limited. Some years ago, power plant operators took notice of the importance of measuring vibrations, as they were proved to be closely related to the state and operating conditions of the machine. Nowadays, monitoring systems are widely used to keep track of the condition of the machine in real time. Sensors are placed in different locations, usually on the bearings, and several parameters are measured constantly in the power plant. This data is then sent to a processing center where trending and diagnostics are made.

Nowadays, many studies are available on how to monitor and diagnose rotating machinery problems using vibration analysis. Most of them are aimed at common machines like pumps and fans with components such as roller bearings and gearboxes [6]. Usually, these are industrially standardized elements, for which common vibration features can be obtained. However, a hydropower plant is more complex: all the unit (turbine and generator) is a tailor-made piece, the installation includes multiple components, and their distributions differ from one power plant to another. Little information can be found about monitoring and damage detection of hydraulic turbines [7][8][9][10][11], and almost nothing about Pelton turbines [12] [13] [14] [15]. The objective of this paper consists in upgrading the current state of condition monitoring of Pelton turbines.

II. CONDITION MONITORING IN PELTON TURBINES

In the beginning of the 90's, vibration monitoring started to be implemented in some hydropower plants with the aim of progressively introducing predictive maintenance. At present, the time intervals between overhauls have been extended and the major problems in the machine are detected before damage is too severe. The general procedure used for monitoring is described in the next paragraphs.

a. Implementation of the system

The first step for vibration monitoring consists in determining the most suitable measuring locations. In a hydraulic machine, like in other rotating machines, the dynamic forces of the rotating train are transmitted to the bearings and their casing, which are also accessible areas. Hence the location selected to place the sensors (accelerometers) was the rigid part of the housing of the main bearings of the machine. In each bearing two orthogonal radial measurement directions were selected on the cap, pedestal or housing. Care was taken to avoid any vibration amplification by local resonances. For horizontal shaft machines, measurements were taken in the vertical and horizontal directions 90° apart (perpendicular to the shaft axis) and in the axial direction. In the case of vertical machine sets, measurements were also taken in all bearings in both the upstream direction and at 90° to that. In all cases the transducers were aligned in all the bearings. Representative locations are shown in **Figure 3** for a horizontal shaft machine. Once the sensors were installed, the vibrations were measured in all the locations at periodic intervals with portable data collectors.

Figure 3. Monitoring locations in a horizontal shaft Pelton turbine

b. Analysis of vibration

The vibration values obtained from the measurements were represented in the frequency domain to characterize the state of the machine. The resulting signal is known as the vibration signature, which depends on the structure and the operating conditions of the turbine (power delivered). Looking at the signature, one can easily detect the frequencies at which the vibration is higher, thus making it easier to identify their causes. In case of wear or damage, other frequencies are excited and the signature varies.

In Figure 4, a typical spectrum of the vibrations measured in a Pelton turbine bearing is shown. In this vibration signature some periodic peaks and random vibrations can be observed. Like in all rotating machinery, the main peaks are at the frequency of rotation f_r and its harmonics (synchronous vibration). These vibrations are generated by forces of mechanical origin (i.e. unbalance or misalignment between the runner and the generator) and forces of hydraulic origin (i.e. the impacts of the water jets on the runner buckets). The frequency at which the jet impinges the structure depends on the rotating speed of the runner and on the number of buckets, and can be defined by:

$$f_p = n \times z_b \times f_r$$

Here, f_p is the bucket passing frequency, Z_b is the number of buckets and f_r is the shaft rotating frequency. n stands for the harmonics of the exciting force, being $n=1,2,3...$

Apart from the synchronous peaks, broad band vibrations can also be identified.

Figure 4. Typical vibration spectrum measured in a turbine bearing

c. Spectral bands

In condition monitoring, the evolution of the vibration levels over time (trend analysis) must be tracked in case there is a dangerous increase. However, it is well known that potentially serious problems can develop within machines, and yet, have a negligible effect on the overall level of vibration. Therefore, spectral bands were selected due to its higher sensitivity to damage detection.

Once the characteristic signature of the machines was determined, the spectral bands were implemented. The spectral bands selected at that time were: a subsynchronous band ($1-0.8 f_r$) for bearing problems, a band around the rotating frequency (0.8 to $1.8 f_r$) to detect unbalance, a band for misalignment (1.8 to $3.8 f_r$), a band for bearing wear and excitation of natural rotor frequencies (3.8 to $f_p-1.8f_r$), a band around the blade passing frequency ($f_p \pm 1.8f_r$) and more bands at higher frequencies. The amplitudes of these spectral bands, overall levels and spectral signatures have been trended and stored in data bases.

With this methodology, condition monitoring has been carried out for several years. This procedure has been very useful to avoid big damage and for diagnostics, but it presents many uncertainties. On the one hand, there is not enough information regarding failures and symptoms in Pelton turbines. Moreover, runner vibrations can hardly be detected in the bearings, because natural frequencies of the runner do not produce important deformations on the rotor. The main challenge consists in knowing how any incipient failures in the turbine can be detected with conventional monitoring techniques, and how the criticality of these can be stated.

Regarding this, one of the limitations in condition monitoring of hydropower plants is the lack of data. In order to detect the symptoms of a specific damage, similar historic cases have to be analyzed previously. Usually, this information is difficult to obtain. Even if there is a data base available, it is not complete enough to describe all types of damage. Apart from that, power plants are not allowed to operate beyond their maintenance limit. This makes it difficult to determine the variation of the monitoring parameters when the turbine is in an advanced degradation state.

For a more accurate monitoring and diagnostic, it is essential to have information regarding failures and symptoms, as well as the vibrations and strains produced by the dynamic forces during operation. The former can be achieved by the analysis of field data obtained by monitoring actual Pelton units. The latter requires creating a model that determines the deformations of the components of the machine and the stresses in the runner.

III. STRATEGY

The procedure that has been used to improve the current state of condition monitoring is represented in Figure 5. Two main parts can be discerned in the whole process: one regarding a comprehensive analysis of the experimental monitoring results and another regarding the development of a theoretical model of actual Pelton turbines.

For the experimental research, all the available Pelton units from the hydropower company were classified depending on the turbine design (head, power, number of jets and number of buckets) and structure characteristics. After that, the data base was analyzed by checking the vibration signatures of all monitored machines, and by looking for the historical cases of damage. For most of the severe failure cases a detailed analysis was carried out in order to identify the symptoms of damage in the vibration signatures.

Figure 5. Procedure to improve condition monitoring

Regarding the theoretical, the first step consisted in creating a numerical model based on Finite Element Method (FEM), which represented the behavior of the most common types of turbines. Geometrical and structural characteristics were provided by the hydropower company. In order to verify the reliability of this representation, the modal frequencies and the mode shapes of the model were compared to the values obtained from on-site tests.

Finally, the validated model was used to simulate the behavior of the machine while being affected by the same operational conditions that, in the past, managed to produce failure on the real machine. The values obtained in the sensor locations of the model were then compared to the information gathered from the data base. Model upgrading was then possible and the distribution of stresses and deformations could be analyzed. At the end, the model could be used to simulate abnormal operational condition that could not be found in historical cases. With this, the symptoms were extracted and the severity of the conditions was assessed.

IV. MONITORING DATA ANALYSIS (FIELD DATA)

a. Data preparation

To tackle the real problem, the experimental data obtained from the monitoring of 28 different Pelton turbines over 25 years has been studied. The Pelton units belong to a hydropower company located in Spain. As mentioned before, real machines are quite complex and the dynamic behavior not only depends on the design of the runner, but also on the structural characteristics (installation type, mounting conditions, environment, etc). To take this into account, the Pelton turbines analyzed have been classified either as machine sets with

a horizontal shaft and pedestal mounted on rigid foundation, or as vertical machine sets with lower bearing housings braced against the foundation.

Firstly, for groups of machines with similar structural characteristics, the vibration signatures were carefully analyzed as to determine the influence of the design parameters and the operating conditions. Secondly, the signatures of the units that suffered similar types of damage were gathered and compared as to find common variations in the vibration patterns. The symptoms for each type of damage were then determined.

b. Types of damage

Pelton turbines can suffer from different types of damage. Most of the cases are due to sand erosion, fatigue or cavitation. Erosion problems are very common in some areas like the Alps and the Andes, where the water carries a large amount of sand particles. In these cases, the most affected locations are the surfaces where the water velocity and/or the acceleration are high. Nozzles, needles and the inner surface of the buckets are usually the most eroded areas [16]. Cavitation can produce pitting on the buckets, especially on the tip and on the cutout lips. Sand erosion enhances the possibility of cavitation because the waviness of the eroded surfaces increases wall turbulence, thus reducing the local pressure. Nevertheless, damage caused by the fatigue of the material is proved to be the most dangerous one [17][18]. The periodic impacts of the water jet lead to a large concentration of stresses at the root of the buckets. After long operation times, these stresses result in cracking the material and destroying the runner buckets. To minimize the effect of fatigue, the design and manufacture of the turbine has to be optimized. At present, reliable runners are constructed using forged blocks of stainless steel due to its improved mechanical properties compared with cast steel, such as fatigue strength and fracture toughness [19]. Even so, mounting and operating conditions can modify the stress distribution in actual runners, thus leading to unexpected failures.

The problems found in the turbines monitored over the years comprise all the types of damage mentioned above. In Figure 6, one can see some examples of the most dramatic cases: a fatigue crack in the tip of a runner bucket, cavitation erosion, a broken bucket and a broken needle. In Figure 6a, a crack developed in the cut-out of a bucket can be seen. Cracks like that can spread quickly and, if unnoticed, can result in a bucket rupture (Figure 6c) with potentially disastrous results.

Figure 6. Some types of damage found: a) Fatigue crack in a bucket; b) Cavitation erosion; c) Runner with a broken bucket; d) Broken needle

c. Symptoms extraction

In this paragraph, an example of the procedure used to correlate damage with symptom is explained. In Figure 7, a typical evolution of the vibration signatures of a Pelton turbine is shown. In the first spectrum of the waterfall one can discern high amplitudes of vibration in

the frequency span of 500-600Hz. This is indicative of an abnormal operation or an incipient damage. Yet, the overall vibration level was of 1.3mm/s rms (root mean square value). According to Standards [20], this machine was still suitable for unrestricted long-term operation. After running under these conditions for some time the vibration increased all of a sudden. This is an evident indicator of damage, so the machine was inspected and one of the buckets was found broken. Further operation would have led to a catastrophic failure. After repair, the frequencies between 500-600Hz didn't appear again in the signatures. At the end, this experience permitted identifying the symptoms tied to this specific type of damage.

The evolution of the signatures was available for all the monitored turbines, so the correlation between symptoms and damage was determined for all types of damage. Here, it is proved that with the analysis of the monitoring data, valuable information can be obtained about the condition of the machine.

Figure 7. Water-fall of vibration spectra measured at different dates

However, to carry out an effective monitoring, each type of damage has to be detected in the initial stage and, if so, its severity determined. The information extracted by looking at the evolution of the signatures is not accurate enough to do so. The symptom described above shows that the natural frequencies were being excited, but the causes leading to this vibration remain unknown. One can only relate the symptoms to the damage, without having knowledge of the underlying dynamics of the turbine. Moreover, the severity of the vibrations of the runner could not be determined with the vibrations measured in the bearing.

The previous example has proved that relying only on field data is not enough to provide an advanced condition monitoring. To obtain a deeper knowledge about the machine dynamics one must resort to numerical models. After these are proved to represent the real machine reliably, they can be used to determine the structural response under different excitation forces and the transmissibility to the measurement spots.

V. NUMERICAL MODEL

This section describes the development of a numerical model of an existing Pelton unit. The machine is a typical horizontal shaft Pelton turbine with two radial bearings and one axial bearing. The objective is to understand how the structural response to a certain exciting force is, and how the vibrations are transmitted to the monitoring locations.

The numerical model was created by means of Finite Element Method (FEM) with the program ANSYS. This comprised the main components of the machine: the runners, the shaft and the alternator. The geometry of the runner was obtained from a 3D scanning process of the real runner, and the shaft and the alternator were reproduced according to the information provided by the company. After that, a mesh was created for the whole machine. The density

of elements was set higher in the areas of the runner's buckets where the concentration of stress is typically high (cut-out and root) in order to obtain maximum accuracy in the results. To check the quality of the mesh, a sensitivity analysis was carried out. Finally, a mesh composed of 1.100.000 elements for the whole machine was selected. In Figure 8, the mesh of the numerical model of the Pelton turbine is displayed.

Figure 8. Physics-based model of the whole machine

To check the validity of the numerical model, the natural frequencies and mode-shapes were calculated and then compared to those of the real machine. To determine the actual modal behavior of the machine an Experimental Modal Analysis (EMA) was done. Natural frequencies were obtained by doing impacts on the runner and on the shaft with an instrumented hammer and measuring the response with accelerometers. The equipment used for the acquisition of signals was a Bruel&Kjaer LAN-XI Pulse system and the accelerometers were Kistler with a sensitivity of (+/- 0,5%). The combination of the hammer signal (input) with the accelerometers signals (output) results in the Frequency Response Function (FRF), which defines the ratio between the vibration levels and the force value. In Figure 9, the comparison between simulation and experimental results can be observed. Runner and rotor modes were then identified and the model was calibrated. The results were compared with the theoretical results obtained from the model.

Figure 9. Identification of modes and comparison between experimental results and the numerical simulation

Once the model was checked and validated, deformations and stresses in the runner for different operating conditions could be computed. The value and distribution of the pressure applied by the jet on the inner surface of the buckets was obtained from previous Computational Fluid Dynamics (CFD) studies [21]. The resulting force was then applied to the FEM model and the response of the structure simulated (Figure 10). The effect of load and head changes could be calculated by changing the velocities and discharge of the jet.

Figure 10. Computation of the dynamic behavior

Finally, with the validated model, the field data obtained during monitoring could be analyzed in more detail for a better understanding of the symptoms. Those cases not found in the data base could be simulated with the appliance of a synthetic damage.

VI. ADVANCED MONITORING

With the methodology described in the previous sections, monitoring could be improved. As shown above, the analysis of the field data indicated that from the types of damage found in Pelton turbines the ones produced by fatigue are the most dangerous.

Fatigue problems generally occur in the runner buckets. The cause can be due to a poor design of the runner, operation at off-design conditions (i.e. higher heads than expected) or problems in the injector, which can produce misalignment between the runner buckets and the water jet. When the jet is centered, only a tangential force is applied on the bucket, yet when this jet is misaligned, the flow on both sides of the bucket is unbalanced and an axial force is also applied. This produces an increase in the axial thrust and modifies the stress levels in the bucket area, which can lead to cracks and bucket failures.

In Figure 11, the vibration signatures of the Pelton obtained during vibration monitoring of the simulated turbine can be observed. The vibrations represented were measured in the radial turbine bearing on different dates. During the inspection of the unit, a few small cracks were found, as well as jet misalignment. In the measured spectra, synchronous vibrations generated by unbalance and the bucket passing frequencies can be identified. It can also be noticed that some natural frequencies of the system were excited.

Figure 11. Vibrations measured during monitoring

The next step consisted in reproducing the vibration behavior of this machine numerically, thus determining the effect of the jet misalignment. The results were compared with monitoring data to see how well the symptoms obtained in the field can be reproduced using the model. The axial force, produced by the jet misalignment, was applied to the numerical model (See Figure 8) and the response in the radial turbine bearing (monitoring position) calculated. In Figure 12, the response of the jet force on the monitoring position (radial bearing) has been represented for a frequency sweep between 0Hz and 300Hz. It can be seen that the maximum response produced by the misaligned jet in the bearing is at frequencies of 40Hz and 120Hz. These are natural frequencies of the rotor whose mode-shapes are represented in the picture.

Figure 12. Response in the radial bearing to the axial forces generated by the jet misalignment.

These numerical results can be compared with the experimental measurements presented in the same Figure. It can be seen that both results agree well and the model is able to reproduce the actual case.

The symptoms detected in the field data (Figure 11) come from the excitation of some natural frequencies at low regions and at a higher frequency span (500-600Hz). The model indicates that the former are rotor natural frequencies and the latter correspond to axial frequencies of the runner, which means the operation of the turbine is not as expected. To see the influence of jet misalignment on the runner vibrations, the response of the buckets, when the jet is aligned and misaligned, was computed (Figure 13). When the jet is aligned with the runner (machine in good condition), the force that is applied to the buckets is tangential and, consequently, the response of the buckets is in the same direction (tangential modes). In case of jet misalignment, rotor and axial bucket modes are also excited (which can be seen in the spectra), thus changing the stress distribution.

With this information the spectral bands for monitoring can be refined, the effects on the stress distribution calculated and the useful life of the runner estimated. With the model other types of damage can also be reproduced.

Figure 13. Deformation of the buckets a) With a centered jet b) With a misaligned jet

Another case of incipient damage generated by jet misalignment in a Pelton turbine is shown in Figure 14. In August 2015 there was a change in the vibration signatures and the excitation of runner axial natural frequencies could also be detected.

Figure 14. Evolution of the vibration signatures in a Pelton turbine. The stress distribution for the runner excited without and with jet misalignment has been represented

With an accurate study of the distribution of stresses in the runner, the possibility of having fatigue damage can be estimated. With this information the owner of the machine can optimize the profit of the machine by adjusting its operating conditions. This can be done by either reducing the load in order to decrease stresses and increase the useful life, or by operating with maximum power in case the price of the energy is high.

VII. CONCLUSIONS

In this paper, a procedure to improve the condition monitoring of Pelton turbines is presented. Firstly, a comprehensive analysis of a data base obtained from vibration measurements of 28 different Pelton turbines over 25 years was done. For the analysis, machines were classified depending on their design and structural characteristics.

After that, the data base was analyzed by checking the vibration signatures of all monitored machines and by looking for the historical cases of damage. For most of the damage cases a detailed analysis was carried out in order to identify the symptoms of damage. By analyzing the vibrations measured before and after repair the symptoms of different types of damage could be extracted.

Of the main types of damage found in the machines (sand erosion, fatigue or cavitation), the most dangerous was produced by fatigue, many times generated by jet misalignment, which can lead to runner buckets to be broken.

Although valuable information was obtained, for a deeper understanding of the generation of these symptoms, a numerical model using FEM method was created. The aim was to simulate the actual dynamic behavior of the whole machine. The model was calibrated with detailed on-site measurements.

With the calibrated model the deformations and stresses generated on the runner could be calculated depending on the operating conditions (head, number of jets, injector opening). The vibrations in the monitoring location could also be calculated.

As an example, the effect by jet misalignment on the runner and the bearing vibrations were calculated using the model. The numerical results agree well with the experimental data. The model can also calculate the change in stress distributions with the abnormal operation of the machine so that problems from fatigue can be estimated.

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REFERENCES

- [1] IRENA - International Renewable Energy Agency (<http://www.irena.org/hydropower>)
- [2] L., Söder, H., Abildgaard, A., Estanqueiro, C., Hamon, H., Holttinen, E., Lannoye, E., Gómez-Lázaro, M., O'Malley, and U., Zimmermann, "Experience and challenges with short-term balancing in European systems with large share of wind power," *IEEE Trans. on Sust. Energy*, vol. 3(4), pp. 853-861, October 2012.
- [3] Angehrn, R., "Safety Engineering for the 423 MW-Pelton-Runners at Bieudron", 20th IAHR Symposium August 6-9, 2000, Charlotte, N.C. USA
- [4] Hermod Brekke "Hydraulic turbines, Design, Erection and Operation" NTNU Trondheim, Norway
- [5] H.Grein, R.Anghern, M.Lorentz "Inspection periods of pelton turbine runners" Proceeding IAHR Symposium, Stirling (UK) 1984
- [6] Tran Van Tung and Bo-Suk Yang "Machine Fault Diagnosis and Prognosis: The State of The Art", International Journal of Fluid Machinery and Systems Vol. 2, No. 1, January-March 2009
- [7] Egusquiza, E.; Valero, M.; Valentin, D.; Presas, A.; Rodriguez, C. "Condition monitoring of pump-turbines. New challenges". *Measurement*. 67, pp. 151 - 163. 05/2015

- [8] C. Rodriguez, B. Mateos, E. Egusquiza "Monitoring of rotor-stator Interaction in pump-turbine using vibrations measured with on-board sensors rotating with shaft" Shock and vibration (2014)
- [9] Egusquiza, E.; Valero, M.; Presas, A.; Huang, X.; Guardo, A.; Seidel, U. "Analysis of the dynamic response of pump-turbine impellers. Influence of the rotor". Mechanical systems and signal processing. 68-69, pp. 330 – 341, 2016.
- [10] Egusquiza, E; Valero, C; Estevez, A; Guardo, A "Failures due to ingested bodies in hydraulic turbines" Engineering failure analysis.18 - 1,pp. 464 - 473.Elsevier, 2011
- [11] E. Egusquiza, C. Valero, X. Huang, E. Jou, A. Guardo, C. Rodriguez "Failure investigation of a large pump-turbine runner" Engineering Failure Analysis.23 (2012) 27-34
- [12] H. Brekke, "Experiences from large Pelton turbines in operation," *Proceedings of the 11th IAHR Symposium Amsterdam*, vol. 2, 1982.
- [13] H.Keck "Commissioning and operating experience with the world's largest Pelton turbines". proceedings Hydrovision 2000 Charlotte (USA)
- [14] M. Egusquiza et al., "Failure investigation of a Pelton turbine runner", *Engineering Failure Analysis*, vol. 81, pp. 234-244
- [15] J. C. Chávez, J. A. Valencia, G. A. Jaramillo, J. J. Coronado, and S. A. Rodríguez, "Failure analysis of a Pelton impeller," *Eng. Fail. Anal.*, vol. 48, pp. 297–307, 2015.
- [16] M. K. Padhy and R. P. Saini, "Study of silt erosion mechanism in Pelton turbine buckets," *Energy*, vol. 39, no. 1, pp. 286–293, 2012.
- [17] J.Schmied, T.Weiss, R.Angehrn "Detuning of Pelton Runners" 7th IFToMM-Conference on Rotor Dynamics, Vienna, Austria, 25-28 September 2006
- [18] H. Brekke, "Safety and Life Time of Hydraulic Turbines," *Proceedings of the 23rd IAHR Symposium*, 2006.
- [19] M. Pradel, "Pelton runner manufacture from forged stainless steel," *Hydropower Dams*, no. January, pp. 25–26, 1995.
- [20] International Standard ISO 10816-5 Mechanical vibration — Evaluation of machine Vibration by measurements on non-rotating parts —Part 5: Machine sets in hydraulic power generating and pumping plants
- [21] C. Vessaz, E. Jahanbakhsh, and F. Avellan, "Flow simulation of a Pelton bucket using finite volume particle method," *Proc. 27th IAHR Symp. Hydraul. Mach. Syst.*, 2014.

Figure1
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Figure2

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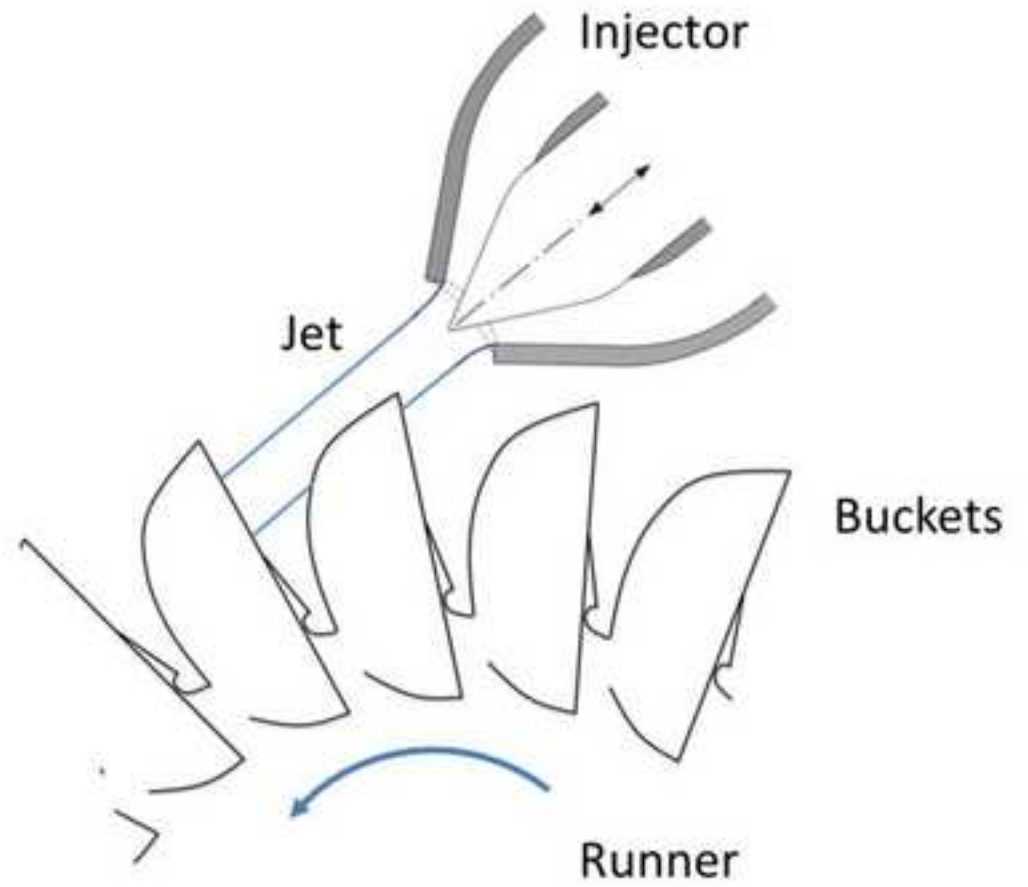
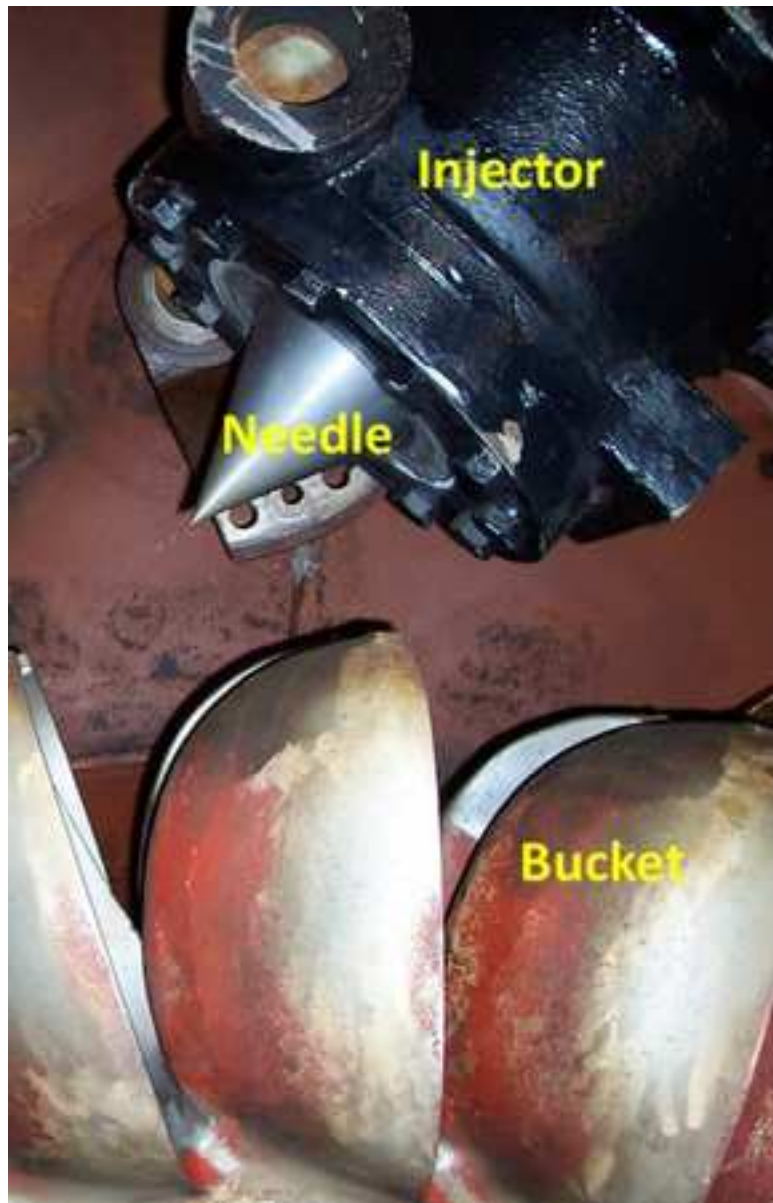


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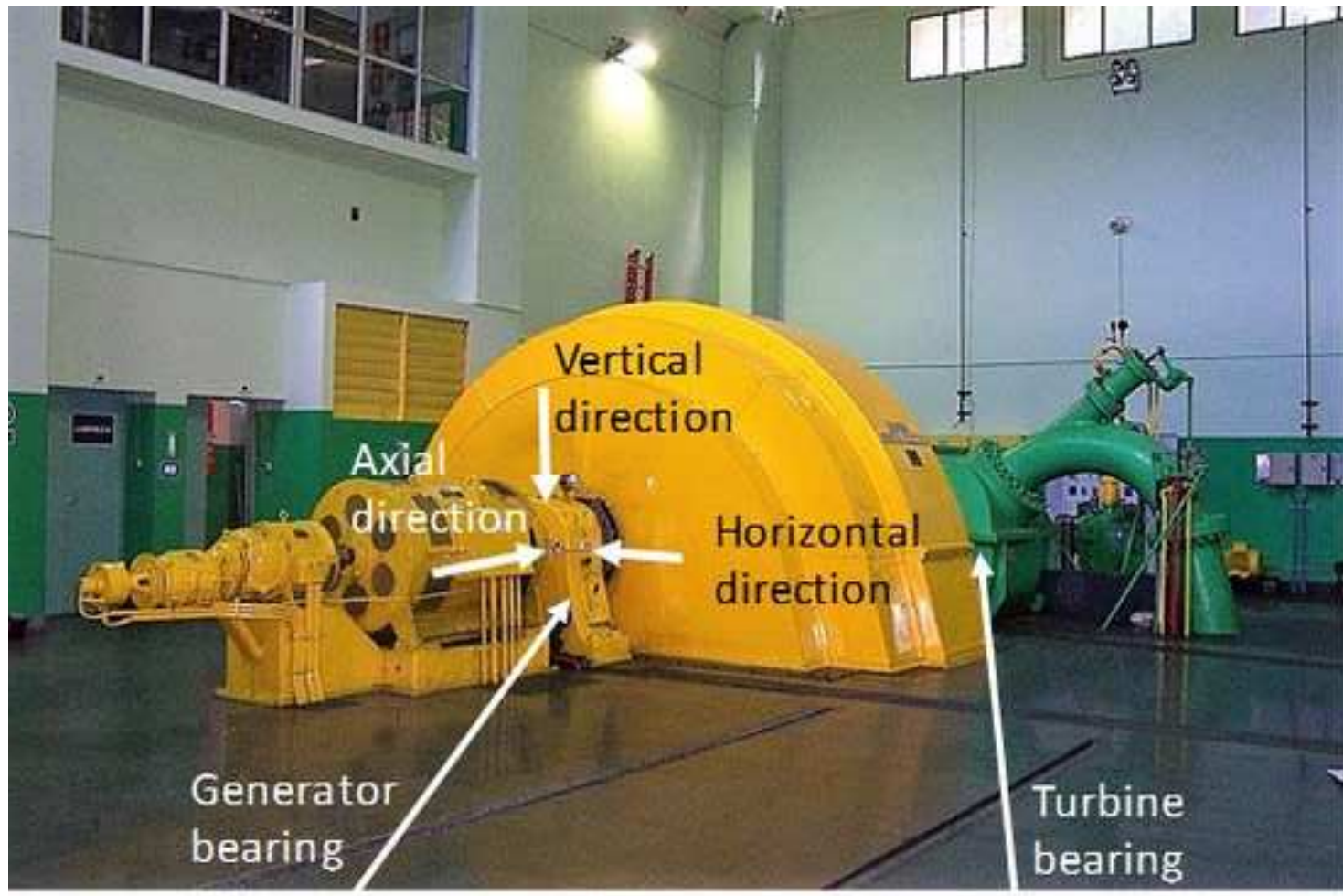


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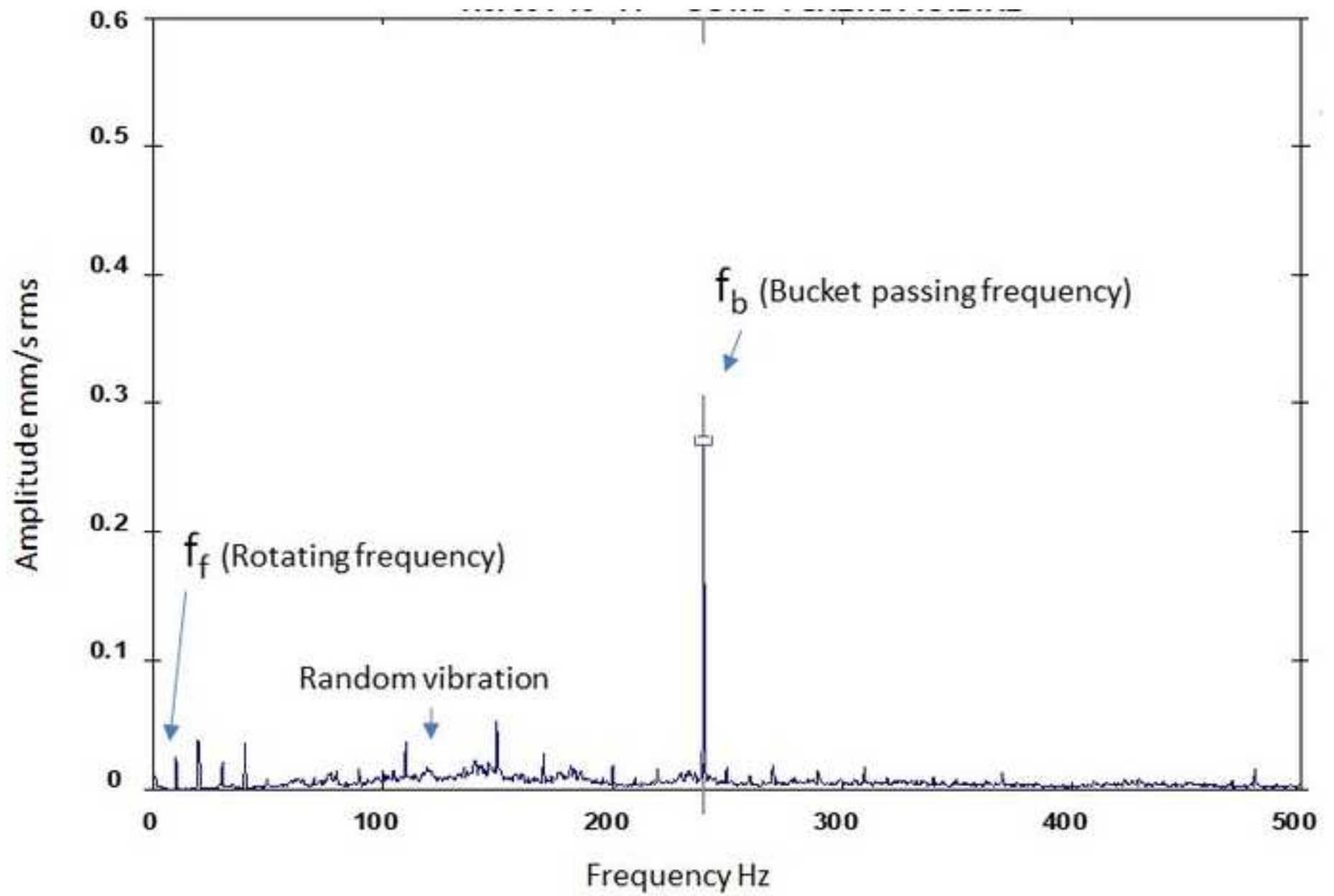


Figure5

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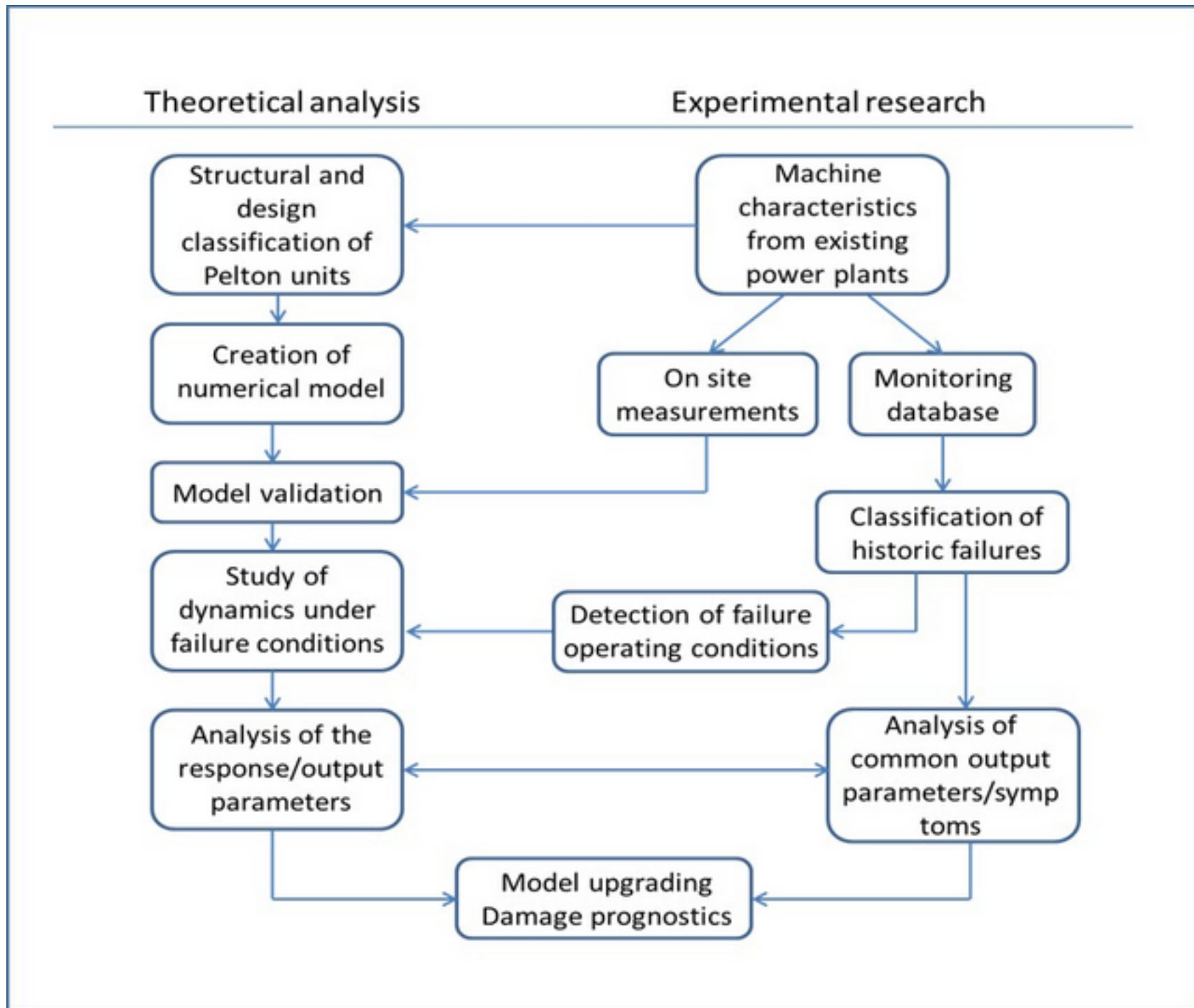


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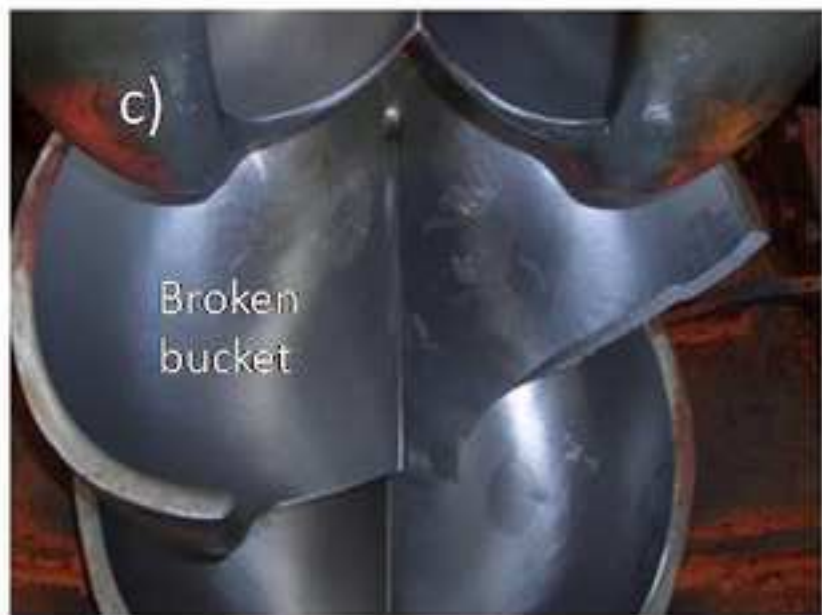
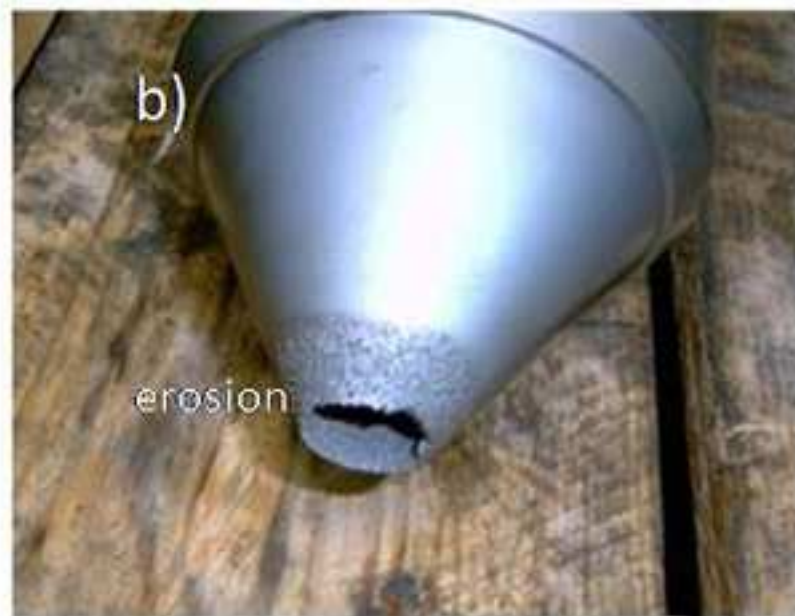
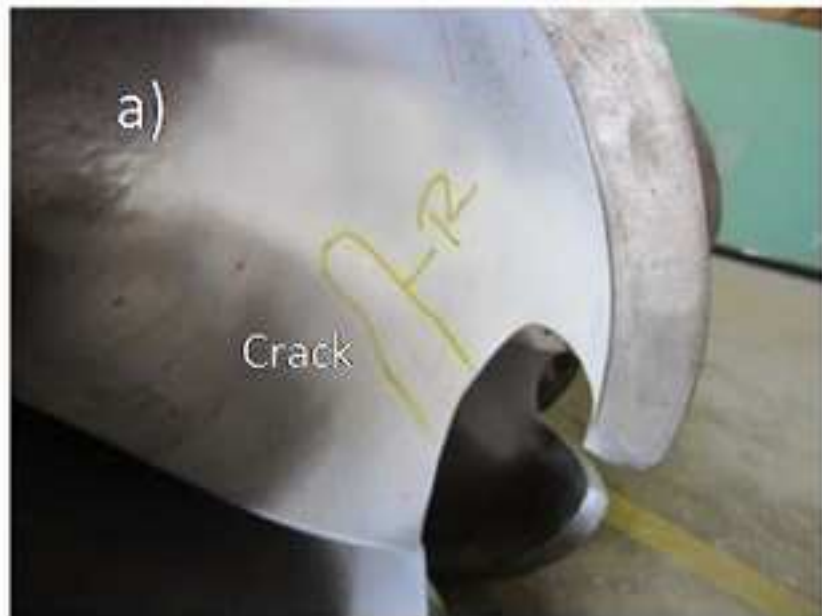


Figure7
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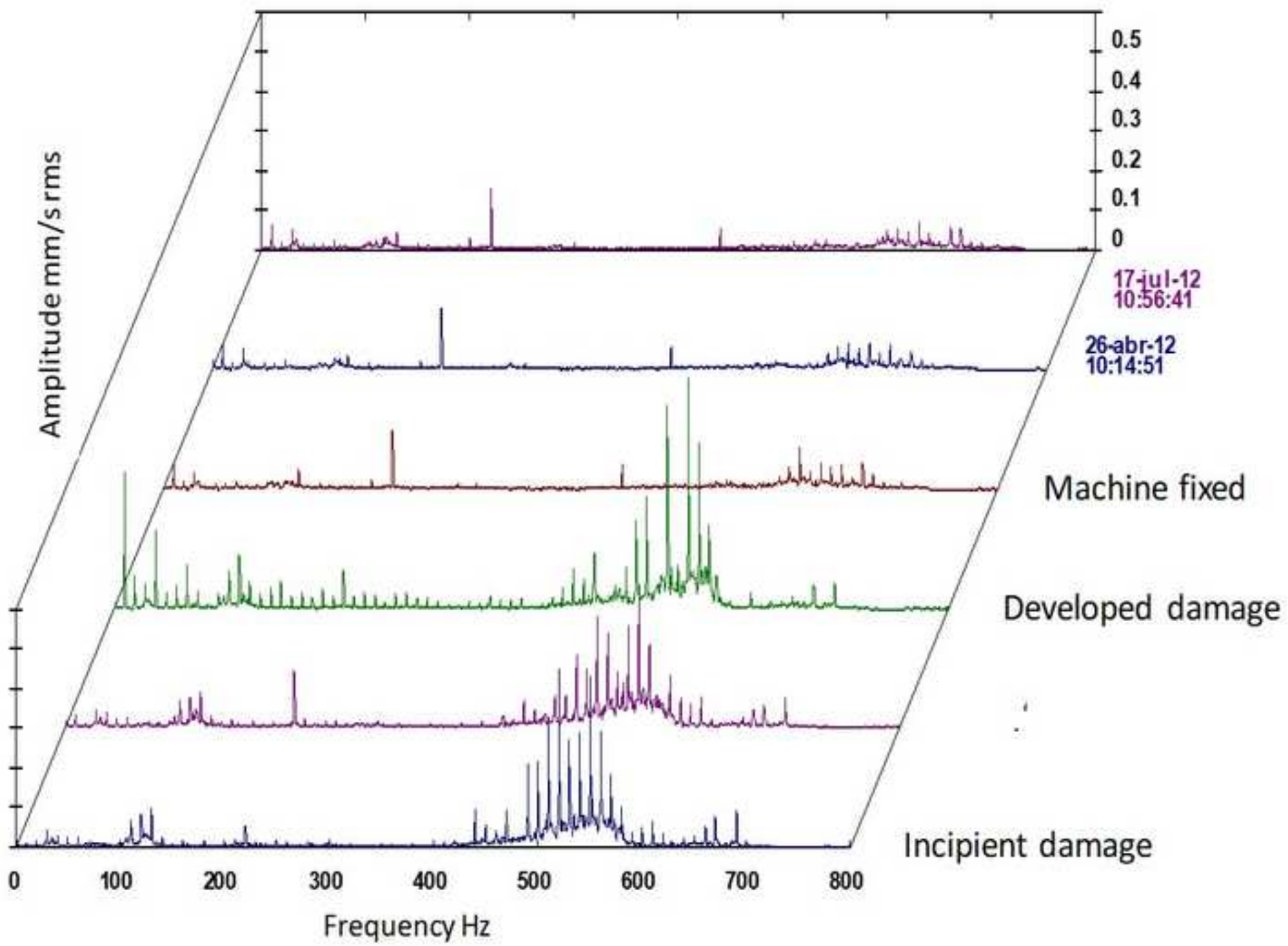


Figure8

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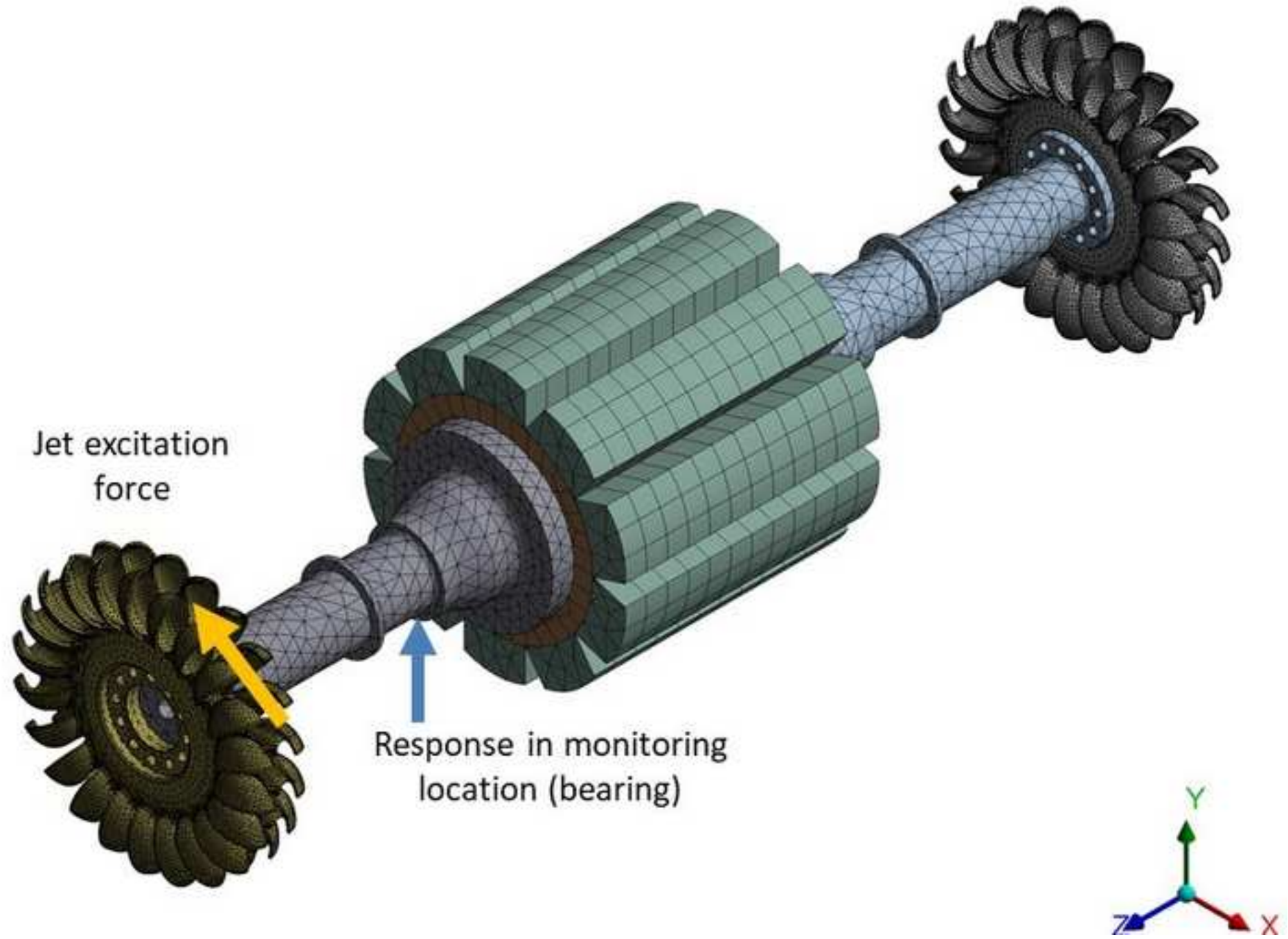


Figure9

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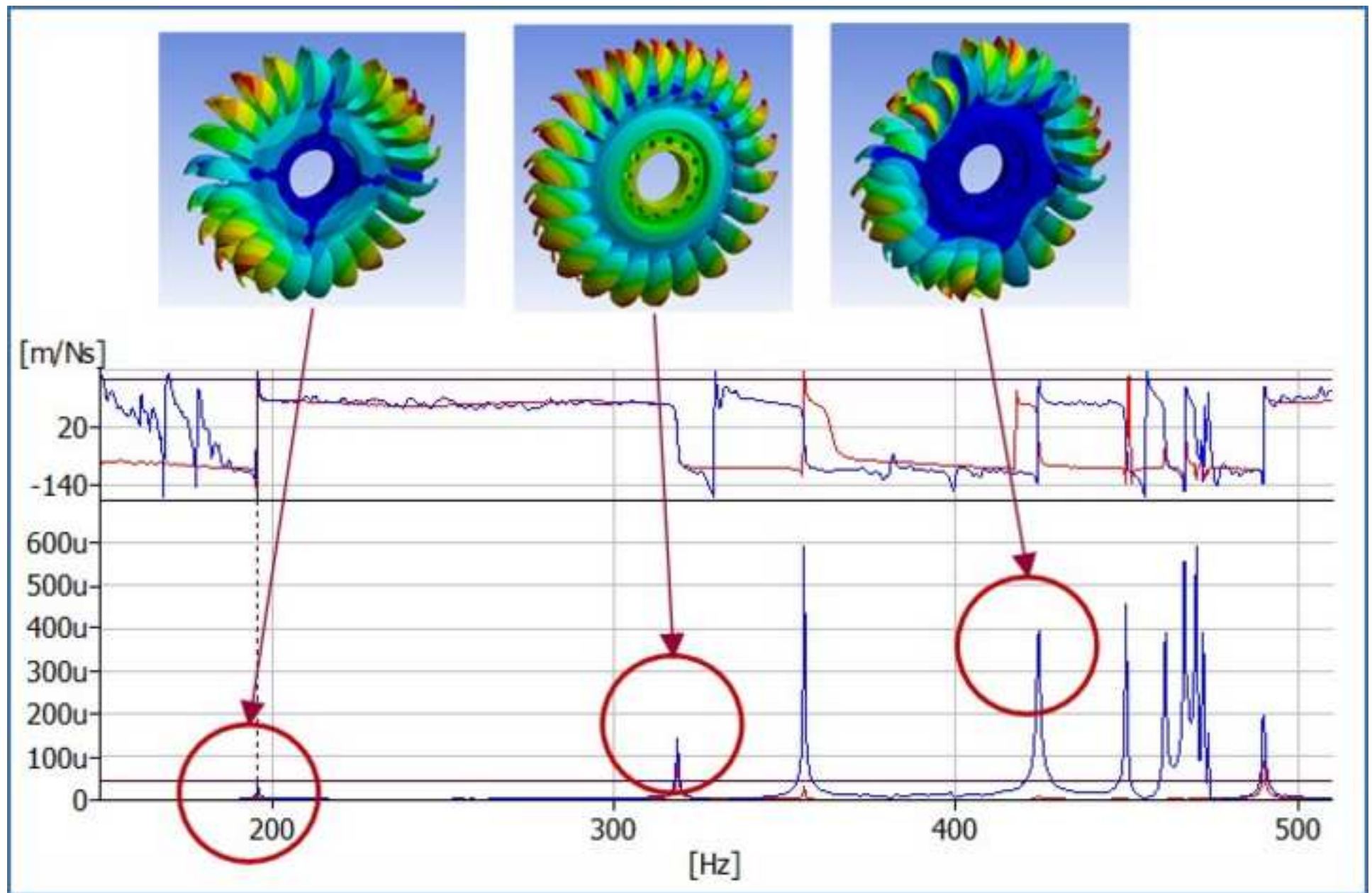


Figure10

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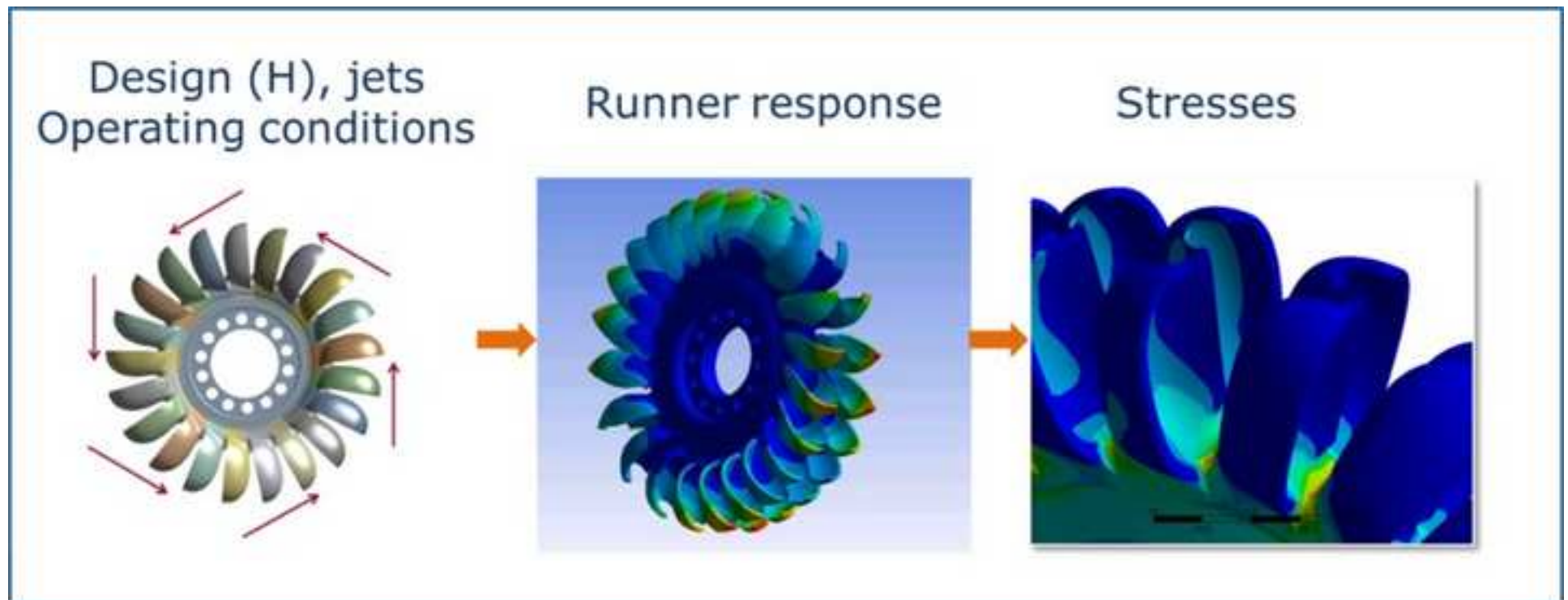


Figure11
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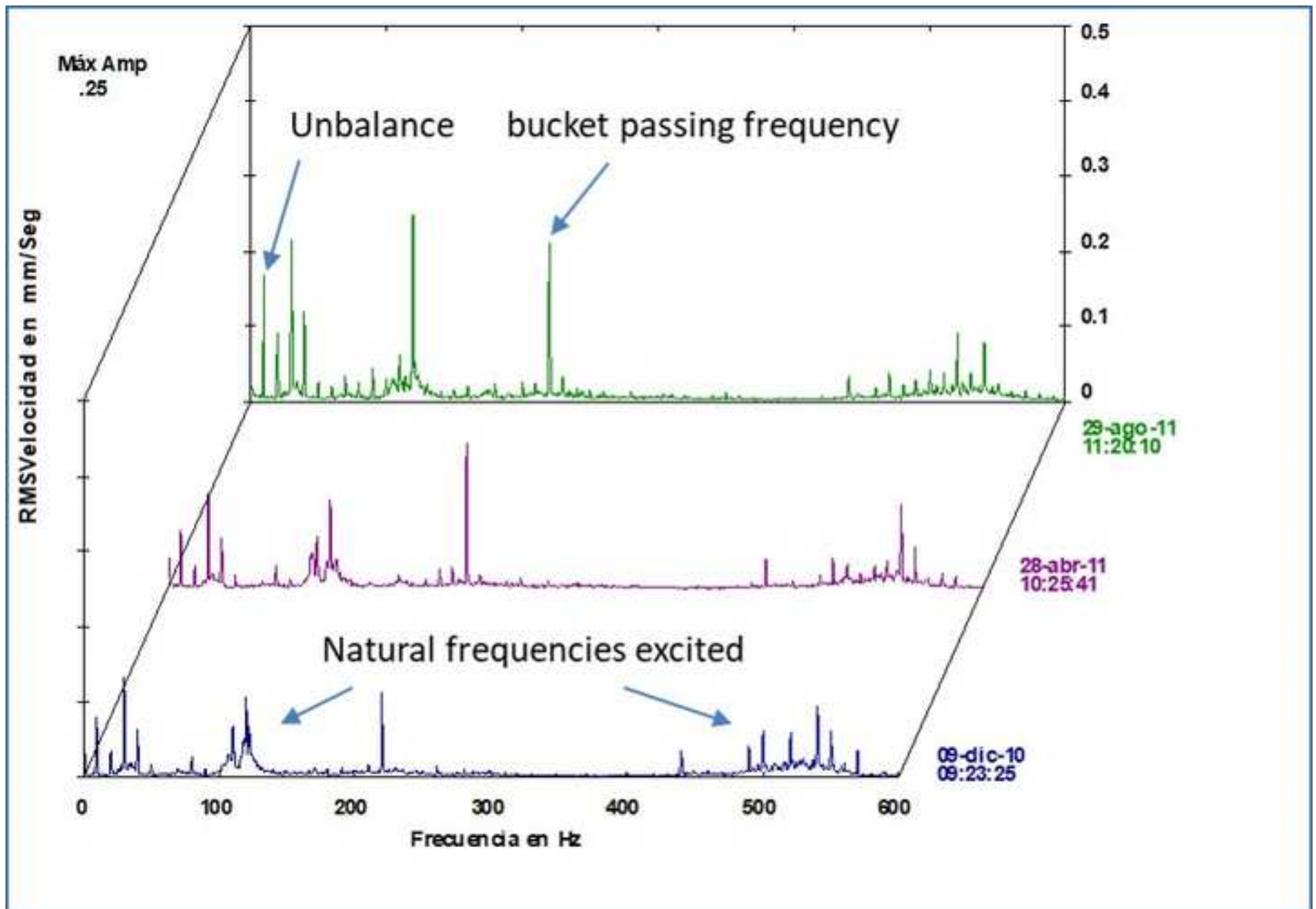


Figure12

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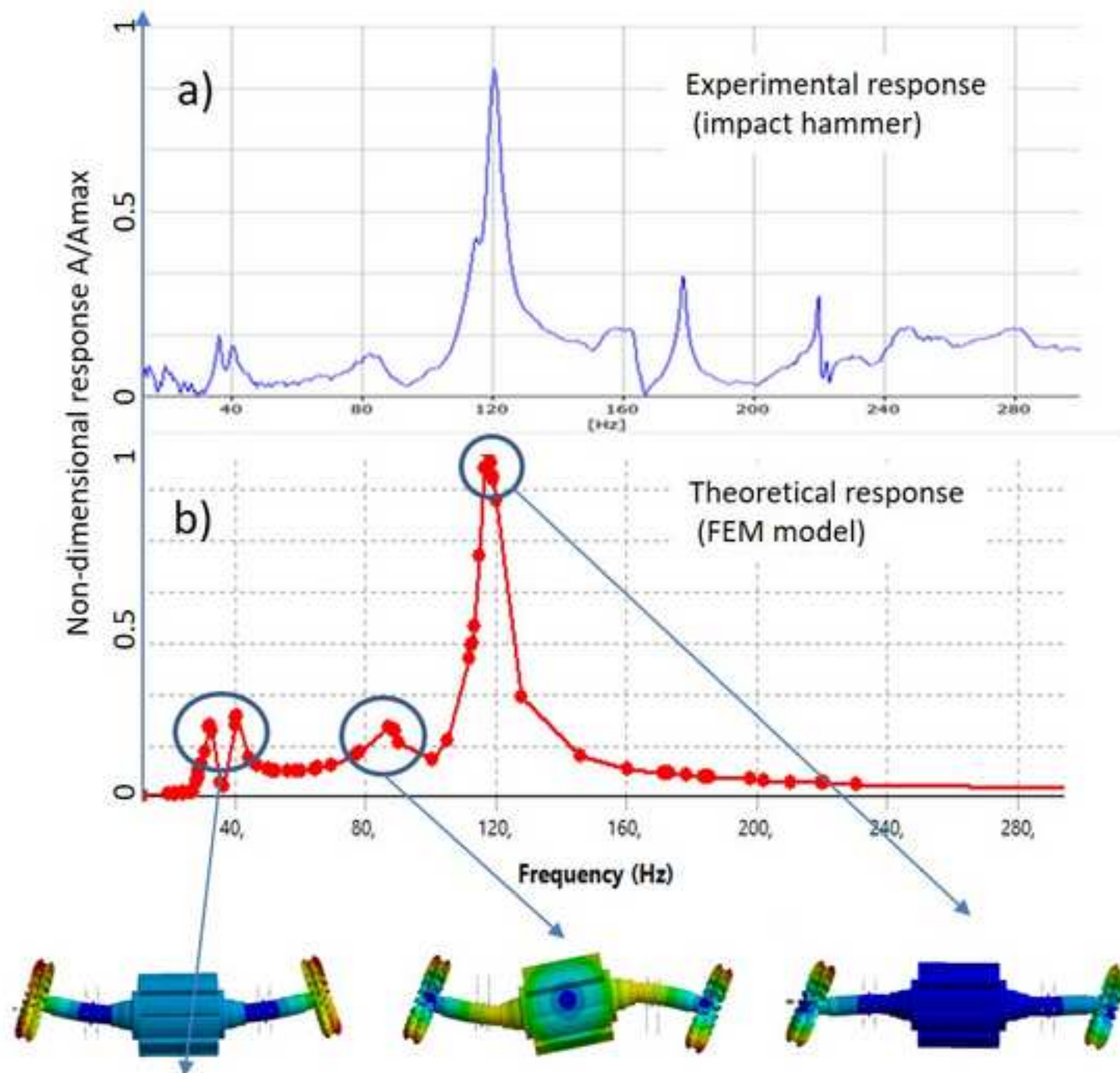


Figure13
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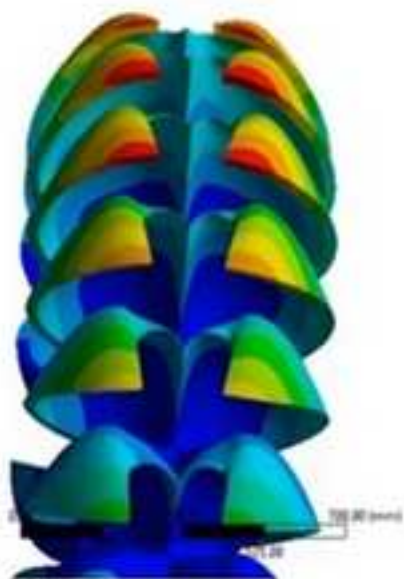
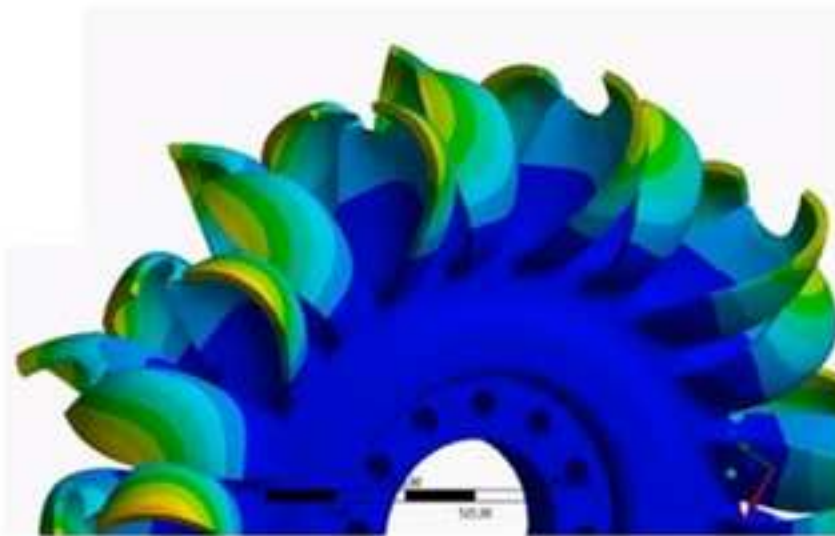
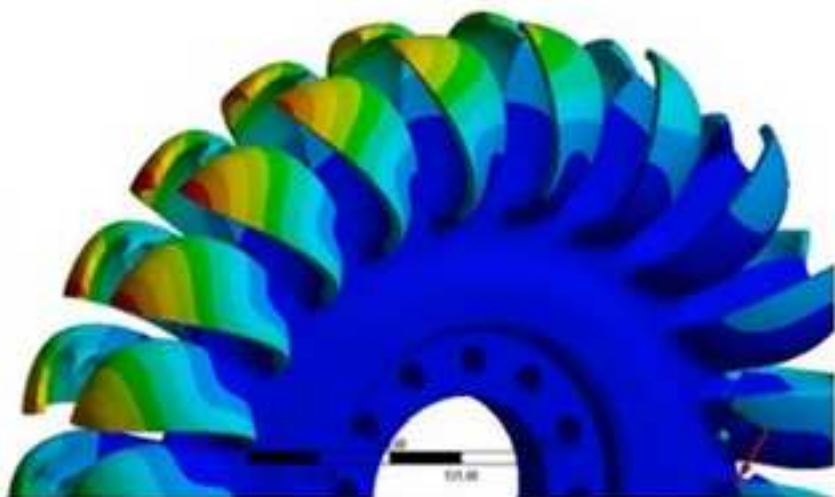


Figure14

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